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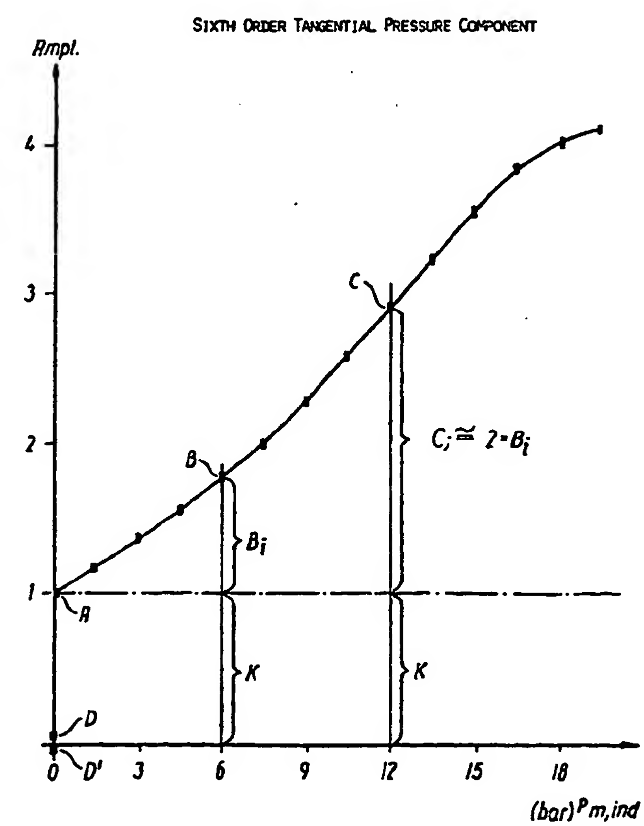
(54) Title: A METHOD OF DIMINISHING EXTRA STRESSES FROM TORSIONAL VIBRATIONS IN A MAIN SHAFT FOR A LARGE TWO-STROKE DIESEL ENGINE

(57) Abstract

A large two-stroke Diesel engine (1) in a ship has a main shaft consisting of a crankshaft and the propeller shaft, and of intermediate shafts, if any. The opening and closing movements of the exhaust valves (12) and the fuel injection into the cylinders are controlled by means of a computer (16). During an interval of revolutions about a resonance point for torsional vibrations in the main shaft the engine is controlled so that the exhaust valve (12) of at least one of the cylinders is set open during the compression stroke of the cylinder, and that the power of at least one of the remaining cylinders is increased corresponding to the lack of power from the cylinder or the cylinders without compression. The contribution (K) of the compression pressure to the vibration order which excites the torsional vibrations is thus removed so that the torsional stresses in the main shaft are diminished.



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A method of diminishing extra stresses from torsional vibrations in a main shaft for a large two-stroke Diesel engine.

The invention relates to a method of diminishing extra stresses from torsional vibrations in a main shaft connected with a propeller for a large two-stroke main engine of a ship, in which the opening and closing movements of the exhaust valves and the fuel injection into the cylinders are controlled by means of a computer, in which the main critical tangential pressure component of the engine causes resonant torsional vibrations in the main shaft in an interval of revolutions wherein continuous running of the engine is prohibited but running-through is necessary, and in which the torsional stresses during the interval of revolutions may exceed a predetermined magnitude.

In a ship, the crankshaft of the main engine is connected with the propeller through a propeller shaft and if necessary one or more intermediate shafts. The crankshaft, the propeller shaft and the intermediate shafts constitute the main shaft of the engine. It is well-known that the engine produces torsional vibrations and thus extra stresses which are superpositioned on the torsional stresses in the main shaft produced by the driving torque, and that the frequency of the torsional vibrations depends on the number of revolutions of the engine. When the frequency of the torsional vibrations for the dominant vibration order, the so-called "main critical order", is close to or coincides with the natural frequency of the main shaft, large torsional stresses may occur in the shaft as a consequence of resonance vibrations. It is also well-known that the natural frequency of the main shaft may lie within an interval of revolutions which is below the point of

maximum continuous rating (MCR) of the engine. The engine therefore has to run through the resonance interval to achieve normal load.

In order to prevent too large extra stresses in said interval of revolutions, the engine may be provided with mechanical vibration dampers. Alternatively, the shaft may be made so rigid that the point of resonance is moved above the point of maximum continuous rating of the engine. Both these solutions are relatively expensive, and the engine plant becomes heavier and larger. The basic concepts in connection with torsional vibrations in the main shaft of an engine and known methods of diminishing such vibrations are described, for example, in the Applicant's brochure "Vibration Characteristics of Two-Stroke Low Speed Diesel Engines", 2nd edition, 1988, by L. Bryndum and S. B. Jacobsen.

The development through the latest years of engines with computer control of the fuel injection and of the valve movements renders possible an individual setting of the injection moment for each engine cycle and for each cylinder. In EP-A-447 697, this has been utilized for suppressing the torsional stresses in the main shaft when running through an interval of revolutions where vibration damping is known from experience to be necessary. The object of this known method of damping the torsional vibrations is to delay the fuel injection in half the cylinders of the engine in order to phase displace the moment contributions of these cylinders in relation to the moment contributions of the other cylinders, whereby the magnitude of the torsional stresses is diminished.

However, by calculations it may be proved that in order to obtain a noticeable reduction, the phase displacement required has to be of such a magnitude that combustion is rendered difficult in practice.

The object of the invention is to provide another and more efficient method of diminishing the torsional stresses in the main shaft.

With a view to this, the method mentioned above is characterized in that the engine during running with increasing load through said interval of revolutions, is controlled so that the exhaust valve for at least one of the cylinders is set open during the compression stroke of the cylinder, and that at least one of the other cylinders is increased in power corresponding to the lack of power from the cylinder(s) without compression.

At the point of resonance for the torsional vibrations in the shaft, the magnitude of the torsional vibrations depends on the magnitude of the individual cylinders' harmonic component of the vibration order which excitates the torsional vibrations, i.e. the so called main critical component. The magnitude of the harmonic component of the individual cylinder depends on both the indicated mean pressure of the cylinder and on the compression pressure in the cylinder. At the numbers of revolutions where torsional vibrations in the shaft usually become large, the compression contribution to the harmonic component is slightly larger than the contribution from the indicated mean pressure. The opening of the exhaust valve during the compression stroke of the cylinder causes both the compression contribution and the mean pressure contribution to be removed, so that the cylinder in question substantially does not contribute with any harmonic component to the torsional vibration. The increased effect on one or more of the other cylinders gives these cylinders a larger mean pressure and thus also larger harmonic components, but as the contribution of the mean pressure to the magnitude of the harmonic components in these cylinders

takes place at the top of the compression contribution, the sum of the harmonic components of the cylinders will be smaller in total. In a six-cylindered engine, the sixth order harmonic components of the cylinders at the point of resonance may, for example, be diminished by almost 7 per cent in this manner by setting one cylinder to be compression-free and letting the other cylinders carry out the work of the compression-free cylinder.

At the passage of the interval of revolutions around the point of resonance, it is possible by means of the method according to the invention to obtain such a diminution of the magnitude of the torsional stresses that there is no need for mechanical vibration dampers or specially designed shaft systems.

The lack of power from the cylinder(s) without compression may be distributed evenly among the other cylinders of the engine. This is especially advantageous, if the point of resonance is so close to the MCR point of the engine that the normal mean pressure of the cylinders at the number of revolutions in question is at such a high level that a single cylinder is not able to yield double of its normal power at that number of revolutions.

In a preferred method according to the invention, at least some of the cylinders of the engine are controlled during said interval of revolutions so that every second cycle they are substantially without compression, and every second cycle yield substantially double the normal power of the cylinder at the number of revolutions in question. The point of resonance will normally be at such a suitable large distance from the MCR point of the engine that the cylinders are able to yield double the power as immediately outside said interval of revolutions. This method of operation offers the advantage that the average thermal load of the

cylinders will be substantially constant during the passage of the point of resonance.

The full effect of the invention may suitably be obtained by controlling the cylinders, seen in the order of firing, to be alternatingly compression-free and double-yielding. Thus, half of the compression contribution of the engine to the resonance vibration is removed.

The exhaust valve which is set open during the compression stroke may suitably be moved to a position at such a limited distance from the valve seat that the outflowing air during the compression stroke is subject to a pressure drop varying with the vibrations at the passage of the valve seat. Thus, a throttle effect over the valve seat occurs, which leads to removal of energy from the system of vibrations which excitates the torsional vibrations. This results in a substantially larger damping of the torsional vibrations than what may be obtained by completely removing the compression contribution of the cylinder, among other things because the pressure of the substantially compression-free cylinder only needs to be relatively few bars above the pressure in the exhaust channel, in order that the amount of energy lost during the throttling has a noticeable effect on the torsional vibrations.

The exhaust valve may also be kept open during the expansion stroke in a position at such a limited distance from the valve seat that the inflowing air is subject to a pressure drop at the passage of the valve seat, whereby two active contributions to the damping are obtained during an engine cycle.

Now, embodiments of the invention will be described in further detail with reference to the very schematic drawings, in which

Fig. 1 illustrates a two-stroke Diesel engine,

Fig. 2 is a diagram of the sixth order tangential pressure component on the connecting rod pin as a function of the mean pressure in the cylinder, and

Fig. 3 shows the sixth order harmonic component of the cylinders in an engine run according to the method of the invention.

Fig. 1 shows a large two-stroke Diesel engine of the crosshead type generally designated 1. The combustion chamber 2 of the engine is delimited by a cylinder liner 3 and a cylinder cover 4 and a piston 5 journalled in the liner.

Via a piston rod 6, the piston is directly connected with a crosshead 7 which is directly connected via a connecting rod 8 to a connecting rod pin 9 in a throw 10 of a crankshaft 11.

An exhaust valve 12 with associated housing 13 is mounted on the cover 4. The exhaust valve is activated hydraulically by a hydraulic drive unit 14 which is controlled by an electro-mechanical valve and activated by control signals transmitted via a wire 15 from a computer 16.

A fuel valve 17 mounted in the cover 4 may supply atomized fuel to the combustion chamber 2. A fuel pump 18 controlled by a solenoid valve and hydraulically driven supplies fuel via a pressure conduit 19 to the fuel valve in dependency of control signals received from the computer 16 via a wire 20. Through a signal-transmitting wire 21, the computer 16 is supplied with information on the current number of revolutions per minute of the engine. The number of revolutions may either be taken from the tachometer of the engine, or it may originate from an angle detector and indicator mounted on the main shaft of the engine and determining the current angular position and rotating speed of the engine for intervals constituting fractions of an engine

cycle of a shaft rotation of 360°. When the computer has determined the time for the fuel injection and the associated amount of fuel, and the opening and closing times of the exhaust valve, the fuel pump 18 and the drive unit 14 are activated accordingly at the moment of the engine cycle which is correct for the cylinder. The engine has several cylinders which are all equipped in the above manner, and the computer 16 may control the normal operation of all cylinders.

If the engine is a stationary power-producing Diesel engine, the main shaft comprises the crankshaft of the engine and, if applicable, a connection shaft to the generator driven by the engine. If the engine is a main engine of a ship, the main shaft, as mentioned above, comprises the crankshaft and the propeller shaft, and any intermediate shafts, if applicable.

The compressive force F_K transmitted via the connecting rod is resolved in Fig. 1 into a radial component F_R and a tangential component F_T. In an interval of revolutions around the natural frequency of the main shaft, the varying tangential forces F_T on the connecting rod pins may produce resonant vibrations in the main shaft. The lowest mode of natural vibration (1-node) of the shaft may receive great energy at a certain number of revolutions from the order of harmonic components of torsional vibrations which corresponds to the number of cylinders, partly because the ignition of the individual cylinders is phase displaced by exactly the period of this vibration order, partly because all the tangential force contributions for this vibration order contribute to intensify the lowest natural vibration of the shaft.

Below, a six-cylindered engine will be described as an example. The relation between the indicated mean pressure in a cylinder and the amplitude for the sixth

order tangential component of vibration oscillations for the cylinder in question is illustrated in Fig. 2. The scaling on the ordinate axis merely indicates the mutual dimensions of the amplitudes, as seen in relation to the amplitude at a mean pressure of 0 bar. When no fuel is injected in the cylinder, the indicated mean pressure is 0 bar, and it may be seen that the magnitude of the sixth order tangential component is found at point A where the amplitude magnitude K is determined by the force occurring on the piston as a result of the compression of the air in the cylinder. The point of resonance for torsional vibrations in the main shaft normally occurs at an engine load where the indicated mean pressure in the cylinders is about 6 bar corresponding to point B on the graph in Fig. 2 where the tangential component is about 80 per cent larger than at point A and is composed of the compression contribution K and the mean pressure contribution B_1. At an indicated mean pressure of 12 bar the amplitude for the tangential component occurs at point C, that is at a value which is about 2.9 times higher than the value at point A. The tangential component C is composed by the compression contribution K and the mean pressure contribution C_1.

Below, an example of the method according to the invention will be described. When the computer 16 determines from the number of revolutions signal received via the wire 21 that the engine is entering the interval of revolutions around the point of resonance for the main shaft, the exhaust valve 12 is activated for at least one of the cylinders to be open during the compression stroke of the piston, so that there will be no build-up of pressure in the combustion chamber. The connecting rod pin 9 is therefore only affected by the inertial forces from the movement of the piston 5, the

piston rod 6, the crosshead 7 and the connecting rod 8, which gives a sixth order tangential component of a very small magnitude, shown at point D in Fig. 2 for the upward piston stroke, and at point D' for the downward piston stroke.

In order to avoid a reduction of the engine power, the computer compensates for the work of the compression-free cylinder(s) by increasing the power of one or more of the other cylinders correspondingly. If a single cylinder, which should have worked at point B in Fig. 2, is set to be compression-free and the computer makes a single other cylinder yield double power, this cylinder will work at point C. Thus, a relative tangential component contribution $K+B_t$ of 1.8 is removed, while the cylinder at point C yields a contribution $K+C_t$ of 2.9, from which may be seen that the tangential components for the two cylinders together will be 0.7 less. As the six cylinders would each have made a contribution of 1.8, if the engine had run in the known manner, compression-free running of a single cylinder causes a decrease in the total tangential component contributions of $(0.7x100)/(6x1.8)\% = 6.5\%$. This reduction may be sufficient to avoid the use of mechanical torsional vibration dampers. If a further diminution of the torsional stresses in the main shaft is desired, two or three of the six cylinders may be run compression-free, which leads to a diminution of about 20 per cent of the sixth order tangential components.

The sixth order tangential component of the individual cylinders is shown in Fig. 3 as a function of the crankshaft angle. As the engine is a two-stroke engine, a full engine cycle corresponds to a crankshaft rotation of 360°. During this cycle, the sixth order tangential components show six full periods. The cylinders 1-4 run at normal mean pressure, while

cylinder 5 runs compression-free, and cylinder 6 runs at double mean pressure. It is seen that the inertia contributions from cylinder 5 outbalance themselves, as they act alternatingly in phase and in counterphase with the tangential component contributions from the other cylinders. As explained above, the total contribution from cylinders 5 and 6 is substantially smaller than the contribution from two of the other cylinders running normally.

When the engine has accelerated out of the interval of revolutions around the point of resonance, the cylinders are controlled in the normal manner.

The interval of revolutions with the point of resonance may be predetermined in dependency of the design of the engine plant in question and may be stored in the computer 16, which then chooses to control according to the method of the invention or in the usual manner depending on whether the actual number of revolutions of the engine is within or outside of the interval. Alternatively, the equipment for measuring the position and speed of revolution of the main shaft may be so sensitive that the computer 16 may determine from the information received concerning the movement of the shaft whether the torsional vibrations of the shaft exceed a predetermined limit value. If this is the case, the computer switches to controlling the engine in accordance with the method of the invention, and when the number of revolutions of the shaft has changed to a predetermined magnitude which is known from experience to be sufficiently far removed from the number of revolutions with too high a level of vibrations, the computer switches back to the normal engine control.

With the above reductions of the total magnitude of the tangential components, it is possible to obtain

a diminution of the extra torsional stresses in the main shaft of up to 65-70 per cent of the undamped level.

From general vibration science it is well-known that the largest value of the amplitude for a harmonic vibration around the point of resonance may be diminished by introducing damping in the vibration system. The torsional vibrations in the main shaft of an internal combustion engine are damped by the frictional forces occurring between the movable parts of the engine. In consideration of the efficiency of the engine, the friction damping is minimized as much as possible. The method according to the invention offers an advantageous possibility of introducing in the vibration system a temporary damping which is only effective during the passage of the point of resonance and thus does not affect the efficiency of the engine at numbers of revolutions outside the resonance interval. This occurs by setting the open exhaust valve(s) in the compression-free cylinder(s) in such a position that the outflowing air is subject to a throttle loss varying with the vibrations at the passage of the valve. The maximum compression pressure of the engine is normally of about 130 bar. If the exhaust valve is set in an open position where the compression pressure in the cylinder only amounts to from 5 to 15 bar, the above compression contribution of the cylinder is negligible in relation to the compression contribution at full compression, but the energy loss at the throttling of the air to the pressure in the exhaust channel produces a damping of the maximum amplitude of the torsional vibrations, which reduces the extra stresses in the main shaft. Thus the extra stresses from the torsional vibrations are diminished, both because the compression contribution is substantially reduced, and because the varying throttling damps the vibrations.

If a larger damping is desired by means of throttling, the exhaust valve may also be kept slightly open during the expansion stroke of the piston, so that gas from the exhaust receiver flows into the cylinder and is throttled over the valve seat. The distance of the valve from the seat need not necessarily be the same for the compression stroke and the expansion stroke, respectively.

PATENT CLAIMS

1. A method of diminishing extra stresses from torsional vibrations in a main shaft (11) connected with a propeller for a large two-stroke main engine (1) of a ship, in which the opening and closing movements of the exhaust valves (12) and the fuel injection into the cylinders are controlled by means of a computer (16), in which the main critical tangential pressure component of the engine causes resonant torsional vibrations in the main shaft in an interval of revolutions wherein continuous running of the engine is prohibited but running-through is necessary, and in which the torsional stresses during the interval of revolutions may exceed a predetermined magnitude,
c h a r a c t e r i z e d in that the engine during running with increasing load through said interval of revolutions, is controlled so that the exhaust valve (12) for at least one of the cylinders is set open during the compression stroke of the cylinder, and that at least one of the other cylinders is increased in power corresponding to the lack of power from the cylinder(s) without compression.

2. A method according to claim 1,
c h a r a c t e r i z e d in that the lack of power from the cylinder(s) without compression is distributed evenly among the other cylinders of the engine (1).

3. A method according to claim 1 or 2,
c h a r a c t e r i z e d in that at least some of the cylinders of the engine (1) are controlled during said interval of revolutions so that every second cycle they are substantially without compression, and every second cycle they yield substantially double the normal power of the cylinder at the number of revolutions in question.

4. A method according to claim 3,

c h a r a c t e r i z e d in that, seen in the order of firing, the cylinders are controlled to be alternatingly compression-free and double-yielding.

5. A method according to any one of the preceding claims, c h a r a c t e r i z e d in that the exhaust valve which is set open during the compression stroke is moved to a position at such a limited distance from the valve seat that the outflowing air during the compression stroke is subject to a pressure drop at the passage of the valve seat.

6. A method according to claim 5, c h a r a c t e r i z e d in that the exhaust valve is also kept open during the expansion stroke in a position at such a limited distance from the valve seat that the inflowing air is subject to a pressure drop at the passage of the valve seat.

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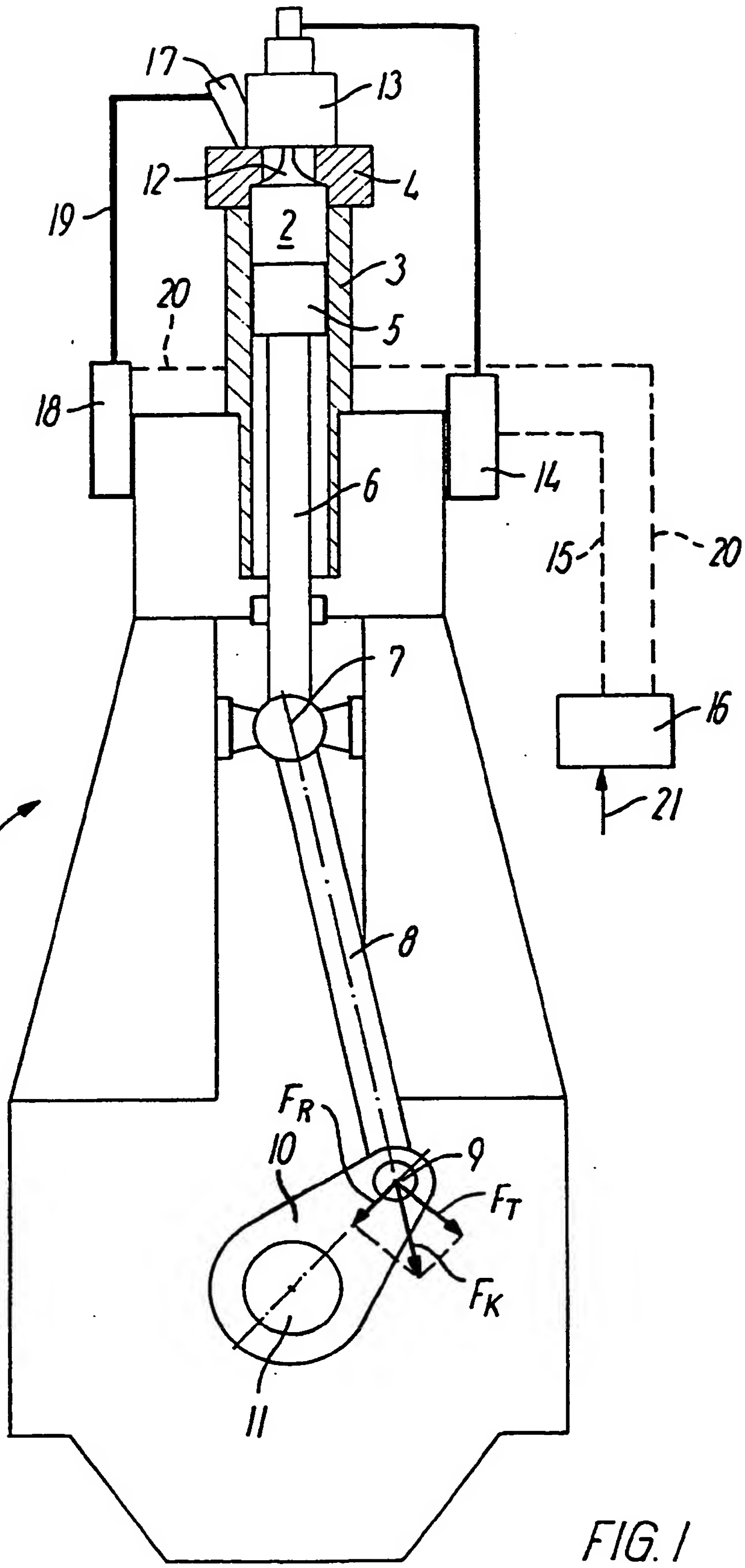

FIG. 1

SIXTH ORDER TANGENTIAL PRESSURE COMPONENT

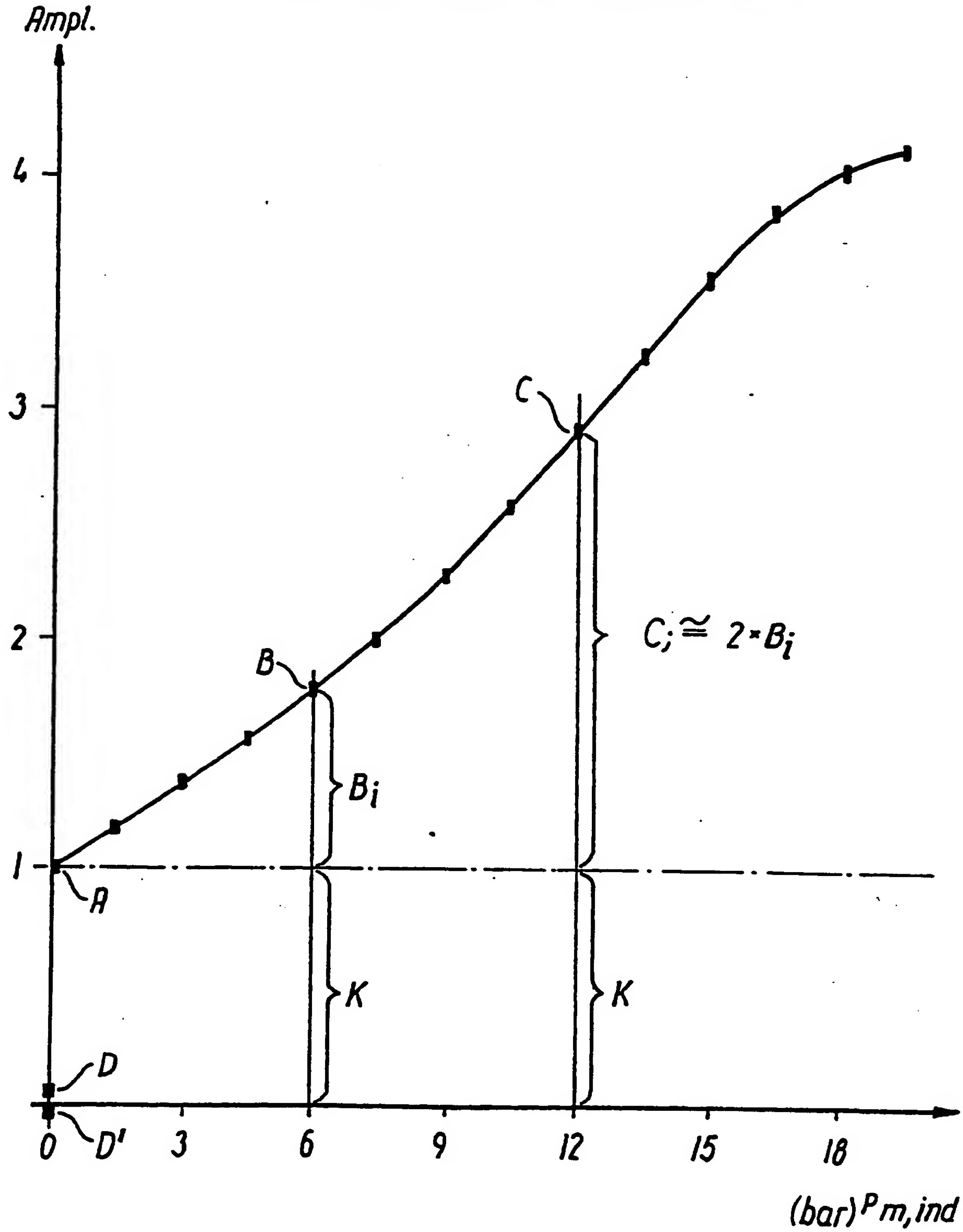

FIG.2

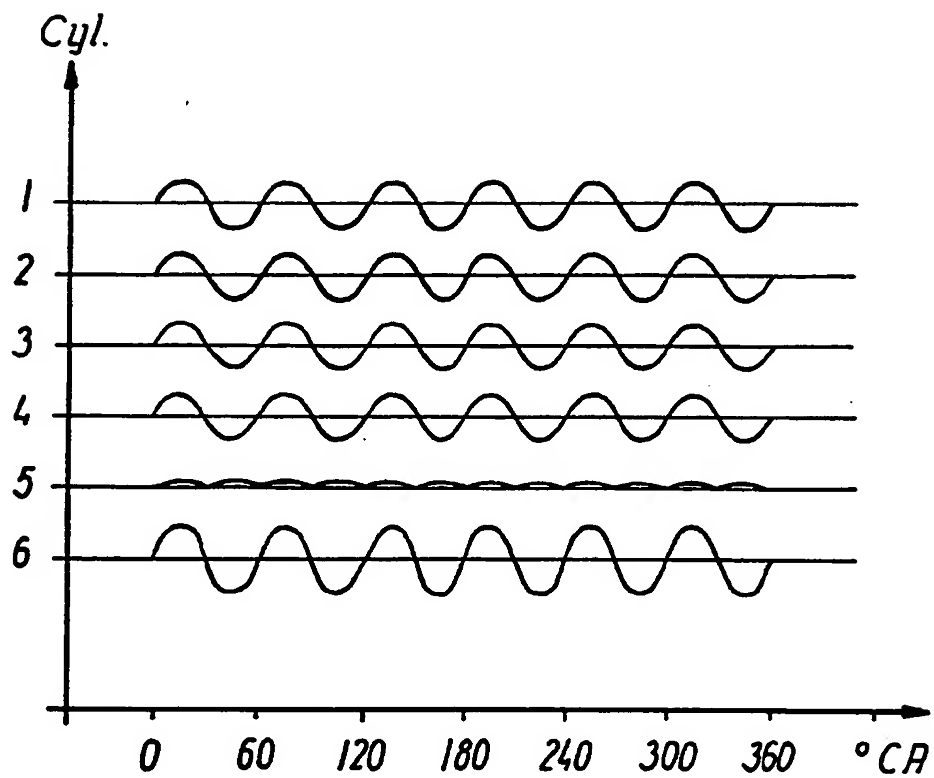

FIG.3

INTERNATIONAL SEARCH REPORT

International application No.
PCT/DK 93/00380

A. CLASSIFICATION OF SUBJECT MATTER

IPC[5]: F02D 13/06, F02D 17/02, F16F 15/20

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

IPC[5]: F02D, F16F

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

SE,DK,FI,NO classes as above

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

CLAIMS,WPI

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	DE, A1, 4126204 (ROBERT BOSCH GMBH), 11 February 1993 (11.02.93), abstract --	1-6
A	DE, C2, 3108374 (NISSAN MOTOR CO. LTD), 28 April 1983 (28.04.83), column 2, line 1 - line 29 --	1-6
A	WO, A1, 9214919 (CATERPILLAR INC.), 3 Sept 1992 (03.09.92), abstract --	1-6
A	US, A, 5090379 (ITO), 25 February 1992 (25.02.92), abstract --	1-6

[X] Further documents are listed in the continuation of Box C. [X] See patent family annex.

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INTERNATIONAL SEARCH REPORT

International application No.
PCT/DK 93/00380

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
A	GB, A, 251238 (ERIC BROWN), 17 March 1927 (17.03.27), page 2, line 39 - line 121	1-6

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INTERNATIONAL SEARCH REPORT

02/07/94

International application No.
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		AU-A- 7772991	15/09/92
		US-A- 5117790	02/06/92
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